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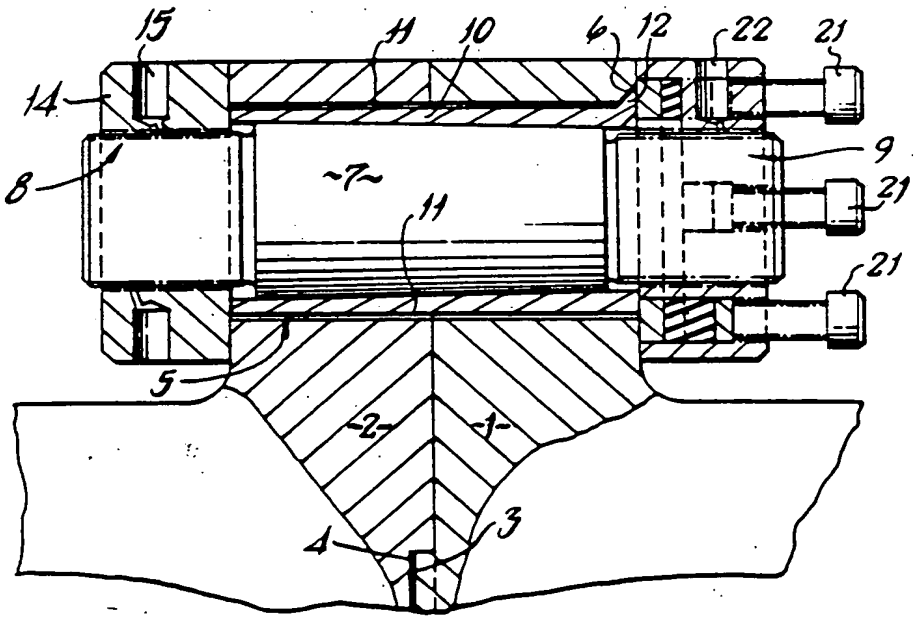

(57) Abstract

An expansion bolt especially for use in flanged couplings in drive shafting comprises a bolt having a tapered shank (7) and a cylindrical sleeve (10) having a complementary internal taper, the sleeve (10) being provided with a pair of diametrically opposed flats (11) in its cylindrical outer surface.

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Bolted coupling.

Field of the Invention

The invention is concerned with flanged connections of the kind used in joining together lengths of power transmission drive shafts. A specific example of such a drive shaft is the propeller shaft of a ship, where very large bending and torsional forces are also accompanied by severe vibration.

The Prior art

The traditional method of dealing with power transmission shaft joints is to use a flanged coupling with a number of equally-spaced stud bolts, each of which is heavily stressed to prevent relative axial movement and to ensure maximum friction between the opposed faces of the flanges. To prevent relative radial movement under shear at least some of the bolts are of necessity "fitted"; they are precisely machined to be a tight force-fit into correspondingly machined holes in the flanges. If there is any relative radial movement, the resultant "fretting" will rapidly cause extensive damage which is both difficult and expensive to repair.

Fitted bolts are difficult to remove; it is often necessary to bore them out, re-machine the bolt holes and then make completely new, oversize bolts, in order to re-connect the coupling. In fact, it is normal to alternate fitted bolts (for shear forces) with plain bolts (for maximum axial force between the flange faces.)

In an attempt to eliminate fitted bolt couplings, it has been proposed to use expanding bolts. These bolts have a slightly tapered shank and fit inside tapered metal sleeves having a complementary internal taper. On tightening the bolt, the respective tapers parts co-operate so that the sleeve is expanded radially with respect to the axis of the bolt to fill the bolt hole through the flanged coupling. An example of such use of an expanding bolt is disclosed in GB-A-1510077.

Discussion of the problem

Marine propeller shaft couplings have by law to be dismantled periodically so that checks for damage can be made. The cost of this work is extremely high because of the time taken to do it. Even with the expansion bolts just mentioned it is not always possible to remove the bolts without damaging the bolt holes and in particular the inner surfaces of the holes. If there is damage and the holes need to be re-machined in situ in the ship, there will be a significant time delay in the dry dock. Such delays can cost over £50,000 a day for a large ship. There will also be the cost of making new, oversize expansion bolts and shells for the re-machined holes in the flanges. The reason for the difficulty of removing expansion bolts is believed to be that the distribution of metal around the holes in a flanged coupling is not symmetrical. Radially-outwards, there is relatively thin metal. Radially-inwards there is the central mass of the coupling. The conventional expansion bolt expands to apply a substantially uniform radially -directed force to the inner surface of the bolt hole, despite the fact that the surrounding material is not equally uniformly distributed. The result is to cause some radially-outwardly directed distortion of the flange material. This generates a bending force in the bolt itself, in addition to the shear forces applied when the coupling is in use. Plastic flow of the metal causes a slightly distorted hole and a bent new line bolt/expansion sleeve. The result is that the bolt and/or sleeve

become jammed in place, or even if they do not jam, the hole may be non-circular and its bore may be damaged by removal of the bolt/sleeve, to the point where a new bolt/sleeve cannot be properly fitted without re-machining the hole and of course, manufacturing a new, oversize bolt/sleeve.

Discussion of the Invention

According to the present invention, an expansion bolt of the kind comprising a tapered bolt and a correspondingly tapered metal sleeve is characterised by the provision of a pair of diametrically-opposed flats on the outer surface of the sleeve. According to a further aspect of the invention, a flanged coupling includes at least two such expansion bolts, the sleeves of each bolt being installed so that the flats thereof are each intersected at right angles by a radius through the axis of the coupling. It will be understood that a "flat" in this present context is a planar surface formed by a machining operation carried out on the previously curved surface of the sleeve.

By installing the sleeve of this invention with the two flats facing respectively radially-inwardly and radially-outwardly of the coupling, the forces generated by tightening the bolt to expand the sleeve are directed substantially circumferentially, along the general line of the bolt (or pitch) circle. This minimizes the risk of jamming, bore damage and bending of the bolt. The force-fit developed (after tightening the bolts) is primarily circumferentially with respect to the flange and this is ideal for resisting the pure shearing load encountered when the coupling is in use to transmit power.

Advantageously, to ensure correct alignment of the sleeves, keying means are provided on each sleeve to correspond with a matching key formed in the flange at or adjacent to each hole into which a sleeve is to be fitted.

In order that the invention be better understood preferred embodiments of it will now be described in greater detail.

Description of preferred embodiments:

The accompanying drawings are as follows:

Figure 1 is a cross-sectional side view through part of an assembled flanged coupling,
Figure 2 is an axial view partly in section through part of the assembled coupling of Figure 1,
Figure 3 is a side view of a stud bolt,
Figure 4 is a cross-sectional side view of an expansion sleeve in accordance with the invention,
Figure 5 is an end view of the sleeve of Figure 4, as seen in direction A, and
Figure 6 is an end view, partly in section of the assembled coupling.

For convenience, like parts in all six figures bear like reference numerals.

Referring firstly to Figure 3, the stud bolt is conventional in that is has a central, slightly tapered shank 7 and screw-threaded end portions 8, 9 respectively. The sleeve 10 best seen in Figures 4 and 5 has an internal taper complementary to that of the stud shank 7. Externally, its cylindrical surface includes two parallel flats 11, one of which terminates in a locating lip 12 formed by bending up a portion of the sleeve towards one end of one of the flats.

Referring now to Figure 1, a flanged coupling in a power transmission shaft comprises a pair of confronting circular flanges 1 and 2 respectively, with a co-operating central alignment recess/projection 3, 4 respectively. The periphery of the flanges includes eight bolt

holes, only one of which is shown in Figure 1, and only three of which are shown in Figure 2 but all of which are indicated in Figure 6.

Figure 1 shows the sleeve 10 inserted into one of the flange holes, with the lip 12 engaged in a matching recess 6 in the flange 1. The bolt is then inserted, from the opposite side, through flange 2, with the shank 3 taper engaged with the matching taper of the sleeve 10. The flats 11 of the sleeve lie in planes perpendicular to the plane of Figures 1, 4 and 5; in Figure 1 they are also intersected at right angles by a radius of the coupling in the plane of the paper. However, this feature is best seen in Figure 2 where the central bolt of the three is shown partly in section to illustrate this point.

In figure 1, the bolt is provided at one end 8 with a circular nut 14 having a number of radial bores 15. These are used to receive the end of a tool such as the cylindrical bar which is commonly used to turn such nuts by hand. The opposite end 9 of the stud bolt is provided with a circular nut 16 of a different kind, although radial bores 22 are still provided for hand tightening. The nut 16 is a hydraulic nut; it comprises an annular cylinder defined in the nut body 16 itself, containing an an annular piston 18 an incompressible fluid mass 19 (typically of rubber) and a piston disc 20. Four equally spaced set screws 21 are provided to apply pressure to the disc 20 and thereby to the fluid mass 19. The construction just described is essentially conventional.

In use, the nut 14 is turned towards the flange 2 until there is a pre-chosen gap between it and the flange. Typicallly the gap will be that equivalent to backing of the nut 14 one complete turn away from the face of the flange. The nut 15 is hand tightened against the flange 1 with the four set screws 21 slackened as far as practicable (as shown in Figure 1). The nut 14 is hand-tightened against the flange 2 and then backed-off through half a turn (180 degrees). The

four set screws 21 are then fully tightened causing the piston 18 to advance against the flange 1, thereby drawing the stud bolt shank further into the taper of the sleeve 10, expanding the latter to firmly grip the inner surface of the bolt hole through both flanges and at the same time to draw the nut 14 hard against flange 2.

It will be appreciated that because of the flats 11 the forces developed against the inner surface of the bolt hole are predominantly directed along the bolt circle (pitch circle) and not radially inwardly/outwardly.

To complete the coupling, every other bolt hole is treated as just described. The intermediate holes are provided with plain shank bolts, to resist axial forces as opposed to torsional (shear) forces, as will now be briefly described.

It will be appreciated that the installation of the intermediate plain shank stud bolts intended to develop axial tension in the coupling is conventional and not quite the same as for the expansion bolts just described. The tension stud bolts are accurately ground to the same length; they are initially firmly tightened by hand after installation. The hydraulic jack nut of each bolt is then used to stress the bolt axially until there is an actual clearance between the nut body and the flange. Shims are inserted and the tensioning set screws are released. The shims prevent the bolt returning to its unstressed length; the actual length under the retained loading is determined by measurement and the stress in each bolt is calculated. If necessary, the shim thickness can be used to alter the retained stress to achieve a calculated, or design value, due allowance being made for losses due to thread deflection/loading.

CLAIMS:

1. An expansion bolt of the kind comprising a bolt having a tapered shank and a cylindrical sleeve having a complementary internal taper characterised in that the cylindrical outer surface of the sleeve is provided with a pair of diametrically-opposed flats.

2. A bolt according to claim 1 further characterised in that the sleeve incorporates keying means whereby the sleeve can be installed with its flats in accurate aligned relation to a particular bolt hole having corresponding, complementary keying means to facilitate such alignment.

3. A flanged coupling for use in between lengths of power transmission drive shafting characterised by the use of at least two of the expansion bolts of claim 1 or claim 2 in said coupling with the flats on the sleeves thereof aligned so as to lie in two parallel planes perpendicular to a radius of the coupling.

4. A flanged coupling according to claim 3 characterised by the use of the expansion bolts of claim 1 or claim 2 in alternate holes of the coupling, the other holes being fitted with plain bolts.

5. A flanged coupling according to claim 4 characterised in that the expansion bolts and/or the plain bolts are provided with hydraulic nuts whereby the bolts may be stressed to a desired extent.

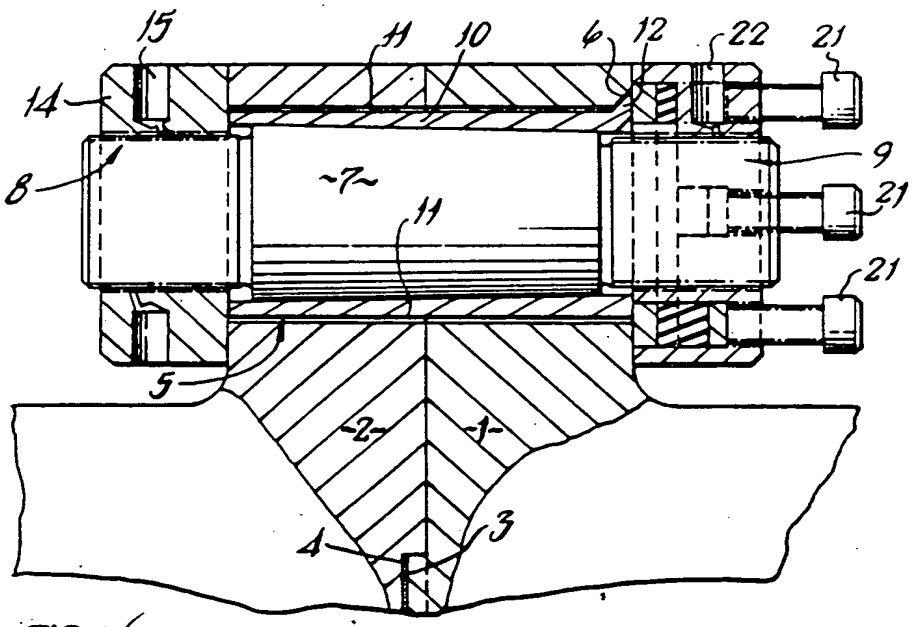
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FIG.1

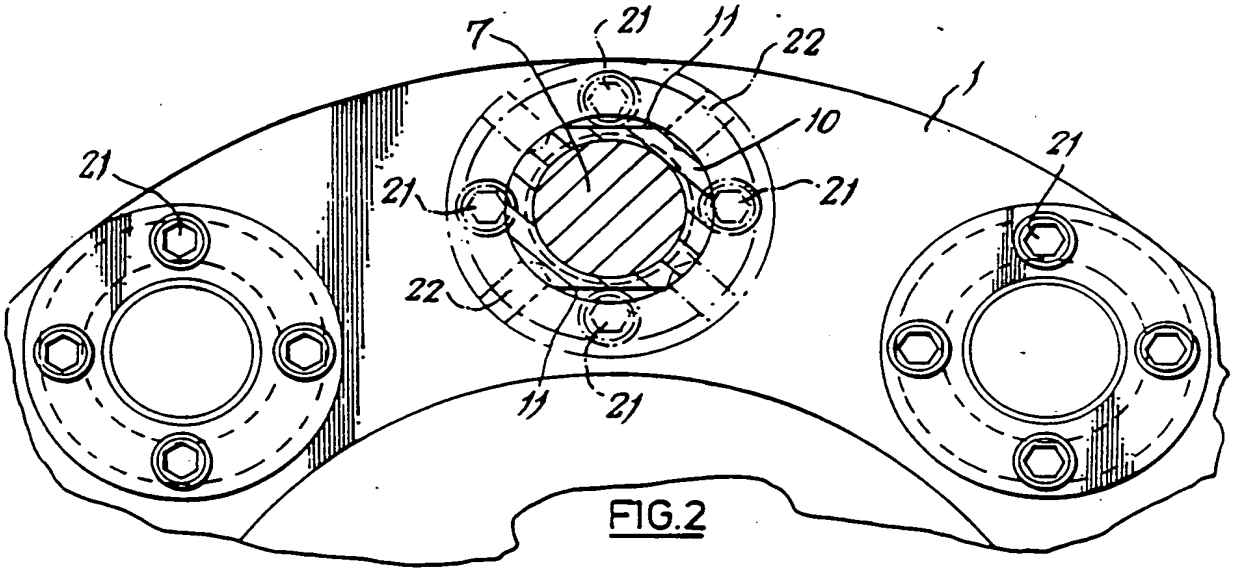
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FIG.2

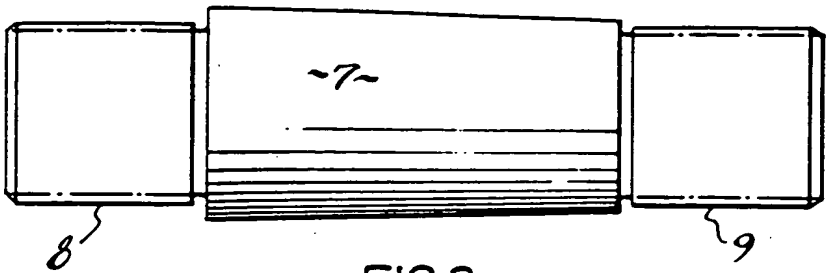

FIG.3

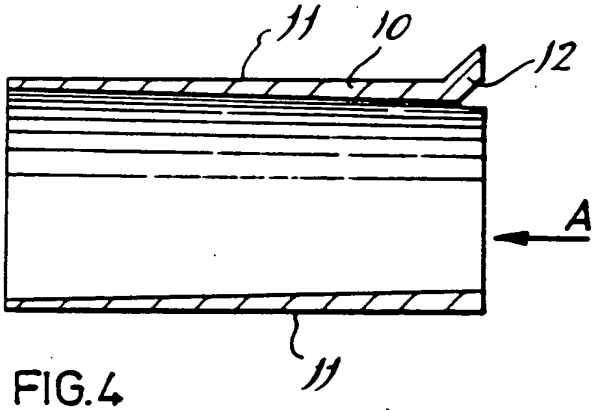

FIG.4

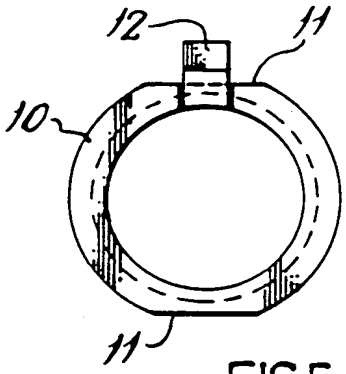

FIG.5

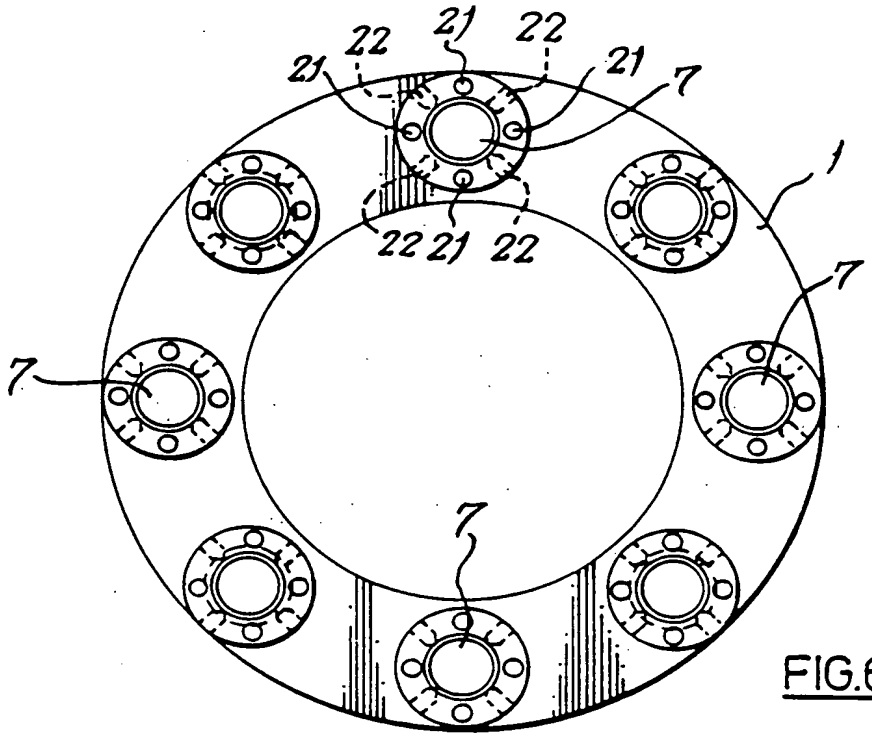

FIG.6

INTERNATIONAL SEARCH REPORT

International Application No PCT/GB 86/00533

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [6]

According to International Patent Classification (IPC) or to both National Classification and IPC

IPC[4]: F 16 D 1/02; F 16 B 5/02//F 16 B 31/00

II. FIELDS SEARCHED

Minimum Documentation Searched [7]

Classification System	Classification Symbols
IPC[4]	F 16 B; F 16 D

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT [9]

Category *	Citation of Document, [11] with Indication, where appropriate, of the relevant passages [12]	Relevant to Claim No. [13]
A	GB, A, 1510077 (ERIKSBERG) 10 May 1978 see page 2, left-hand column, lines 6-25; figures 1,2 cited in the application	1

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IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
12th May 1987	- 5 JUN 1987
International Searching Authority: EUROPEAN PATENT OFFICE	Signature of Authorized Officer: M. VAN MOL

ANNEX TO THE INTERNATIONAL SEARCH REPORT

INTERNATIONAL APPLICATION NO. PCT/GB 86/00533 (SA 14522)

This Annex lists the patent family members relating to the patent documents cited in the above-mentioned international search report. The members are as contained in the European Patent Office EDP file on 25/05/87

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Patent document cited in search report	Publication date	Patent family member(s)		Publication date
GB-A- 1510077	10/05/78	BE-A-	845592	16/12/76
		NL-A-	7607172	01/03/77
		FR-A-	2322296	25/03/77
		DE-A-	2637756	10/03/77
		JP-A-	52038634	25/03/77
		SE-B-	389076	25/10/76

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